RECEIVER-DRYER FOR IMPROVING REFRIGERATION CYCLE EFFICIENCY

CROSS-REFERENCES TO RELATED APPLICATIONS

[0001] Not applicable.

FEDERALLY SPONSORED RESEARCH OR DEVELOPMENT

[0002] Not applicable.

REFERENCE TO A MICROFICHE APPENDIX

[0003] Not applicable.

BACKGROUND OF THE INVENTION

Field of the Invention

[0004] The present invention generally relates to automotive air conditioning systems. More specifically, this invention is directed to a receiver-dryer for use in an automotive air conditioning system wherein the receiver-dryer includes unique features for improving the efficiency of the separation of a gas phase from a liquid phase of a refrigerant fluid and for redirection of the liquid phase so as to improve sub-cooling of the refrigerant through the receiver-dryer and a condenser.

Description of the Related Art

[0005] Air-conditioning systems for motor vehicles are well known. Figure 5 illustrates an example of a typical air-conditioning system 10, which essentially includes a compressor 12, a condenser 14, a thermal expansion valve 16, an evaporator 18, a refrigerant line 20 connecting the aforementioned components

together, and a refrigerant fluid flowing therethrough (as represented by the various arrows). It is also known to provide a receiver-dryer 22 in a refrigeration circuit between the condenser 14 and the thermal expansion valve 16 to remove particulates and moisture from the refrigerant fluid and thereby protect the downstream components.

[0006] At the beginning of a refrigeration cycle, an upstream side 24 of the compressor 12 receives a gaseous phase of the refrigerant fluid. Powered by an engine of the motor vehicle (not shown) via a belt drive 26 and clutch 28 or electrically driven system, the compressor 12 compresses the refrigerant fluid to increase the temperature and pressure to create a superheated vapor and to pump the refrigerant downstream through the refrigerant line 20 to the condenser 14.

[0007] Within the condenser 14, the superheated refrigerant fluid changes from its gaseous phase to a mostly liquid phase. The superheated vapor of the refrigerant fluid flows through interior passages 30 of the condenser 14 while ambient air flows over exterior surfaces 32 and cooling fins 34 of the condenser 14. The superheated vapor is much hotter than the ambient air. Thus, the heat of the superheated vapor is given off to the surrounding ambient air flowing over the exterior surfaces 32 and fins 34 of the condenser 14, thereby cooling the refrigerant fluid in accord with heat transfer principles. As the refrigerant fluid continues to flow through the condenser 14 and lose more heat to the surrounding ambient air, it begins to condense from its gaseous phase into a liquid phase. Eventually, the refrigerant fluid exits the condenser 14, mostly in a liquid phase (X) but typically including some gaseous portion, and flows downstream through the refrigerant line 21, and enters the receiver-dryer 22.

[0008] The receiver-dryer 22 includes an adsorbent unit 36 therein for dehydrating or removing water from the refrigerant fluid. The receiver-dryer 22 includes an outlet line 38 having a pickup end 40 disposed in a lower region 42 for communicating only liquid phase, and not gaseous phase, refrigerant out of the receiver-dryer 22 and downstream to the thermal expansion valve 16.

[0009] The thermal expansion valve 16 "expands" the refrigerant fluid so as to suddenly reduce the pressure of the refrigerant fluid. This sudden reduction in pressure causes the refrigerant fluid to be sprayed through the refrigerant line 20 downstream to the evaporator 18.

[0010] Within the evaporator 18, the evaporation process extracts the required evaporator heat from an incoming stream of fresh or recirculating interior air, thereby cooling the air. The now latent heat of liquid fluid phase of the refrigerant fluid changes back into a gaseous phase as a result of the heat received from the fresh or recirculating interior air. While the now relatively cool refrigerant fluid flows through interior passages (not shown) of the evaporator 18, relatively hot ambient air flows over exterior surfaces (not shown) of the evaporator 18, in similar fashion as the condenser 14. The evaporator 18 cools the hot moist ambient air because the humidity or water vapor in the hot ambient air collects or condenses on the exterior surfaces of the evaporator 18. The evaporator 18 also dehumidifies the hot moist ambient air because the moist ambient air is given off to the relatively cold refrigerant flowing through the evaporator 18, thereby warming the refrigerant fluid and cooling the air flowing over the exterior surfaces of the evaporator 18. Thus, a supply of cool, dry, dehumidified air flows away from the evaporator 18 and into a passenger compartment of the motor vehicle (not shown), while the heated gaseous refrigerant

flows out of the interior passages of the evaporator 18, through the refrigerant line 20 downstream back to the compressor 12 where the refrigeration cycle repeats.

[0011] Referring to prior art Figures 5 and 6, there is shown a pressure vs. enthalpy diagram of the prior art refrigeration cycle with pressure depicted along the ordinate and enthalpy depicted along the abscissa. Schematic points O, A, D, and F of Figure 5 are graphically represented in Figure 6 as points O, A, D, and F of the refrigeration cycle. In general, path O-A represents the compression stage of the refrigeration cycle, path A-D represents the condensing stage, path D-F represents the expansion stage, and path F-O represents the evaporation stage of the refrigeration cycle. Point B represents the transition point at which the refrigerant condenses from a superheated vapor to a saturated vapor. Point C represents the transition point at which the refrigerant further condenses from a liquid-vapor mixture to a saturated liquid.

[0012] In prior art air-conditioning systems, under vehicle usage conditions there may - or may not- be sub-cooling at the output side (range X – in Figure 5, B-C) of the condenser (14 in Figure 5), depending upon the state of the refrigerant fluid due to various vehicle performance variables. In other words, and referring to Figure 6, range X represents the variable nature of the refrigerant fluid temperature at the downstream or output side of the condenser 14 at range X in Figure 5 and Y_1 represents the sub-cooling of prior art refrigeration cycle. Whereas point A is well defined and fixed at the location on the pressure vs. enthalpy diagram as shown, range X is not so well defined and varies along the condenser path A-D of the pressure vs. enthalpy diagram depending upon the vehicle performance variables of vehicle speed and load on the air-conditioning system. The slower the vehicle speed, or at idle

condition and, the higher the load on the air-conditioning system, the sub-cooling range Y_1 diminishes and may approach zero. Under these conditions, the refrigeration cycle looses sub-cooling capability and operates only in the "X" range. Likewise, point D is dependent upon the amount of sub-cooling that can be performed on the refrigerant beyond point C. In other words, point D is incrementally dependent upon the cooling load and quantity of ambient air flow when the air conditioning system is properly charged with refrigerant.

[0013] Referring to Figure 6, the amount of heat (Q) that can be removed by the condenser (14) is represented by the equation $Q=M_{R134a} * (h2-h1)$. M_{R134a} is the variable mass flow for R134a refrigerant while h2 is the enthalpy at the beginning of the refrigerant entering into the condenser, 14 and h1 is the enthalpy at the receiver dryer outlet D. Assuming a constant mass flow, the greater the range in enthalpy that the air-conditioning system can produce, the greater the heat that can be removed.

[0014] More recent advancements in automotive refrigeration suggest structurally integrating a receiver-dryer with a condenser. For example, U.S. Patent 5,927,102 to Matsuo et al. teaches a receiver that is integrally mounted to a condenser in such a manner as to maintain a constant sub-cool temperature. The '102 patent discloses the condenser as including a pair of opposed and vertically extending first and second header tanks and a core composed of a plurality of tubes extending between the header tanks in a generally horizontal fashion. At the top of the first header tank, an inlet joint is disposed into which superheated refrigerant from the compressor flows. At the bottom of the second header tank, an outlet joint is disposed out of which substantially condensed refrigerant flows. Inner spaces of the header tanks are divided by separators into an upper space into which the superheated

refrigerant flows and a lower space into which flows refrigerant cooled down in the core. The receiver is mounted to the condenser in fluidic communication between the upper and lower spaces of the condenser. More specifically, the receiver-dryer is mounted to the condenser such that the receiver does not overlap with the upper space in order to minimize heat transfer from the incoming superheated refrigerant to the refrigerant fluid collected in the receiver, thereby minimizing evaporation of the refrigerant fluid. Accordingly, a "whole" space of the receiver can be reserved for adding make up refrigerant to compensate for loss of refrigerant due to leakage, while maintaining a constant sub-cool temperature.

[0015] From the above, it can be appreciated that receiver-dryers of the prior art are not fully optimized. For example, while the '102 patent does teach passive stabilization of the sub-cooling temperature of the condenser, it does not teach active optimization of sub-cooling of the condenser. In other words, the '102 patent focuses on passively avoiding evaporation of the liquid phase of the refrigerant fluid within the condenser, rather than actively maximizing condensing of the gas phase into the liquid phase. Moreover, the performance of the prior art receiver-dryer of Figures 5 and 6 is excessively dependent upon vehicle operating conditions and air conditioning demand. Thus, there remains a need for an integrated receiver-dryer that is less dependent upon vehicle operating conditions and air conditioning demand, and that not only minimizes evaporation of a liquid phase therein, but also maximizes the liquid phase so as to return relatively more liquid phase to the condenser for additional sub-cooling, thereby enabling the condenser to consistently output 100% sub-cooled liquid phase refrigerant.

BRIEF SUMMARY OF THE INVENTION

[0016] The present invention contemplates a receiver-dryer for use as part of an integrated receiver-dryer-condenser of an air-conditioning system of an automotive vehicle, wherein the receiver-dryer optimizes or maximizes a liquid phase of refrigerant therein so as to return relatively more separated liquid phase to a condenser for additional sub-cooling of the refrigerant.

[0017] According to the preferred embodiment of the present invention, there is provided a receiver-dryer including a substantially cylindrical vessel having an interior defined by a base wall, a side wall extending vertically upwardly from the base wall, and a concave end terminating the side wall and disposed substantially opposite of the base wall. A refrigerant inlet pipe extends into the interior of the vessel in a generally vertically upward direction and terminates in an exit end that faces the concave interior end of the vessel. The refrigerant inlet pipe is adapted for directing refrigerant as a liquid and gas mixture into contact with the concave end such that the refrigerant impinges on the concave end to disperse the refrigerant into a total gaseous phase that accumulates in the upper portion of the vessel and a liquid phase that runs down the interior surfaces of the concave end and side wall of the receiver-dryer for cooling and for accumulation in the lower portion of the vessel. A refrigerant outlet pipe is in fluidic communication with the interior of the vessel.

[0018] In another aspect of the present invention, an integrated receiver-dryer-condenser is adapted for use in air conditioning system, wherein the integrated receiver-dryer-condenser includes a condenser and a receiver-dryer fluidically connected to the condenser.

[0019] The condenser of the receiver-dryer-condenser includes a first vertically disposed header tank, a second vertically disposed header tank spaced substantially laterally opposite of the first vertically disposed header tank, and a core positioned between the first and second vertically disposed header tanks. The core includes a plurality of horizontally disposed passages in fluidic communication with the first and second vertically disposed header tanks for communicating refrigerant fluid therebetween. An inlet is disposed in one of the first and second vertically disposed header tanks and is adapted for receiving a superheated gaseous phase of the refrigerant fluid. An intermediate outlet port is disposed in one of the first and second vertically disposed header tanks and is adapted for exiting a mixture of a gaseous phase and a liquid phase of the refrigerant fluid. An intermediate inlet port is disposed in one of the first and second vertically disposed header tanks and is adapted for receiving a dispersed liquid phase of the refrigerant fluid. An outlet is disposed in one of the first and second vertically disposed header tanks and is adapted for exiting a sub-cooled liquid phase of the refrigerant fluid.

[0020] The receiver-dryer of the integrated receiver-dryer-condenser includes a substantially cylindrical vessel having an interior defined by a base wall, a side wall extending vertically upwardly from the base wall, and a concave end terminating the side wall. A refrigerant inlet pipe is disposed in fluidic communication with the intermediate port of the condenser, extends therefrom into the interior of the vessel in a generally vertically upward direction, and terminates in an exit end facing the concave end. The refrigerant inlet pipe is adapted for directing refrigerant into contact with the concave end such that the refrigerant impinges on the concave end to disperse the refrigerant into a gaseous phase that accumulates in the upper portion of

the vessel and a liquid phase that runs down the interior surfaces of the concave end and side wall for heat transfer cooling and for accumulation in the lower portion of the vessel. A refrigerant outlet pipe is disposed in fluidic communication with the interior of the vessel and with the intermediate inlet port of the condenser.

[0021] In a further aspect of the present invention, a method is provided for sub-cooling refrigerant within an air conditioning system. The method includes receiving a superheated high pressure gaseous phase of a refrigerant fluid in a condensing stage of a condenser and condensing the superheated high pressure gaseous phase of the refrigerant fluid therein into a mixture of a gaseous phase and a liquid phase. The method further includes communicating the mixture into a vertically disposed vessel and directing the mixture into an upper concave surface of the vertically disposed vessel, thereby dispersing the liquid phase from the gaseous phase wherein the liquid phase falls toward a lower portion of the vessel over a desiccant material, and further thereby cooling the gas and liquid phases for improved sub-cooling of the liquid phase and for improved condensing of the gas phase into the liquid phase. Finally, the method includes communicating the now separated, cooled, and dehydrated liquid phase out of the vessel.

[0022] It is an object of the present invention to provide an improved receiver-dryer for use in an improved integrated receiver-dryer-condenser of an automotive air-conditioning system and to provide an improved method of sub-cooling refrigerant within an automotive air-conditioning system.

[0023] It is yet another object to provide an integrated receiver-dryer that is less dependent upon vehicle operating conditions and air conditioning demand placed

on an automotive air-conditioning system, compared to prior art receiver-dryer designs.

[0024] It is a further object to provide a receiver-dryer that is capable of not only minimizing evaporation of a liquid phase of refrigerant therein, but is also capable of maximizing the liquid phase therein so as to return relatively more liquid phase to a condenser for additional sub-cooling.

[0025] It is still a further object to provide an integrated receiver-dryer-condenser that outputs 100% sub-cooled liquid phase refrigerant fluid.

[0026] It is yet a further object to provide a more simplified and cost effective integrated receiver-dryer-condenser that is at least as efficient as prior art designs.

[0027] These objects and other features, aspects, and advantages of this invention will be more apparent after a reading of the following detailed description, appended claims, and accompanying drawings.

BRIEF DESCRIPTION OF THE SEVERAL VIEWS OF THE DRAWINGS

[0028] Figure 1 is a schematic view of a refrigeration system according to an embodiment of the present invention, illustrating a condenser and a receiver-dryer according to an embodiment of the present invention;

[0029] Figure 2 is a pressure vs. enthalpy diagram illustrating the refrigeration cycle of the refrigeration system of Figure 1;

[0030] Figure 3 is a cross-sectional view of the receiver-dryer of Figure 1;

[0031] Figure 4 is a cross-sectional view of a receiver-dryer according to an alternative embodiment of the present invention;

[0032] Figure 5 is a schematic view of a refrigeration system according to the prior art; and

[0033] Figure 6 is a pressure vs. enthalpy diagram illustrating the refrigeration cycle of the prior art refrigeration system of Figure 5.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

[0034] Generally shown in the Figures, an integrated receiver-dryer-condenser is provided within a refrigeration system in accordance with an embodiment of the present invention for improved refrigerant sub-cooling and refrigeration cycle efficiency. A receiver-dryer of the integrated receiver-dryer-condenser is designed to optimize or maximize a liquid phase of refrigerant therein so as to return relatively more liquid phase to a condenser of the integrated receiver-dryer-condenser for additional sub-cooling.

[0035] Referring now in detail to the Figures, there is shown in Figure 1 a refrigeration system 110, which operates in accordance with a method of the present

invention. The refrigeration system 110 generally includes the following components: a compressor 112 for compressing and pumping refrigerant through the condenser 116; an integrated receiver-dryer-condenser (IRDC) 114 having, mechanically attached, a condenser 116 for condensing the refrigerant into liquid, and a receiver-dryer 118 for separating and cooling the refrigerant; a thermal expansion valve 120 for expanding the refrigerant; an evaporator 122 for evaporating the refrigerant into gas; and a refrigerant line 124' and 124" for communicating the refrigerant among the aforementioned components. The compressor 112, thermal expansion valve 120, and evaporator 122 may be of conventional design, manufacture, and composition that is typical for such refrigeration system components.

[0036] The compressor 112 is mounted within an engine compartment of a motor vehicle (not shown) such that the compressor 112 is powered by an accessory drive belt 126 that connects to a crankshaft pulley of an engine (not shown) or is electrically driven (not shown). Rotation of the engine translates into rotation of the compressor pulley to power the compressor 112 when a clutch 126 on the compressor 112 is engaged. Accordingly, the compressor 112 suctions gaseous refrigerant from an upstream portion of the refrigerant line 124" into an inlet port 130 thereof, compresses the gaseous refrigerant into a high pressure, high temperature superheated gaseous state, and pumps the refrigerant out an outlet 132 downstream toward the IRDC 114. Referring to the pressure vs. enthalpy diagram of Figure 2, this compression process is represented by path O-A.

[0037] Referring again to Figure 1, the condenser 116 of the IRDC 114 generally includes a pair of opposed header tanks defined by a first header tank 134

and a second header tank 136, and further includes a heat exchanging core 138 fluidically connected between the header tanks 134, 136. The core 138 includes a plurality of horizontal tubes or passages 140 having opposed ends in fluidic communication with the header tanks 134, 136. Corrugated cooling fins 142 are disposed between exterior surfaces 144 of the passages 140 for cooling the refrigerant flowing therethrough. The header tanks 134, 136 are basically vertically disposed hollow vessels having horizontal partitions, dividers, or separators D1-D5 therein. The first header tank 134 includes an inlet port 146 and the opposite, second header tank 136 includes an outlet port 148. It is contemplated, however, that one or the other of the header tanks 134, 136 could include both the inlet and outlet ports 146, 148 depending upon how many and in what location the horizontal partitions D1-D5 are used. Thus far described, the condenser 116 is preferably composed of aluminum, is manufactured in accordance with known condenser manufacturing techniques, and is designed in accord with typical condenser design configurations, with the below-mentioned exceptions.

[0038] Preferably, five separators D1, D2, D3, D4, D5 are used to divide the condenser 116 into sub-sections. A condensing stage of the condenser 116 is defined between the inlet port 146 and the fifth separator D5, and a sub-cooling stage is defined between the fifth separator D5 and the outlet port 148. The fourth and fifth separators D4, D5 are disposed at the same elevation within their respective header tanks 136, 134, such that there is no fluidic communication between the condensing and sub-cooling stages within the condenser 116 itself. A person skilled in the art will recognize that the number of separators used is a function of the application and

therefore the five separators as disclosed in the preferred embodiment is not intended to be limiting. Any number may be used, or adapted for the application

[0039] However, the receiver-dryer 118 of the IRDC 114 fluidically communicates the condensing stage of the condenser 116 to the sub-cooling stage of the condenser 116. The receiver-dryer 118 communicates with an intermediate outlet port 150 at the end of the condensing stage of the condenser 116 via an inlet tube, stand pipe, line 152, or the like, that extends centrally and upwardly within a generally cylindrical housing 154 and terminates in an exit end 156 in an upper portion 158 of the housing 154. An integrated filter and adsorbent unit 160 is mounted about the inlet line 152 for dehydrating or removing water from the refrigerant. An outlet line 162 extends downwardly from a lower portion 164 of the housing 154 and communicates through an intermediate inlet port 166 with the sub-cooling stage of the condenser 116. The inlet and outlet lines 152, 162 are preferably brazed or joined mechanically to the housing 154 and connected to the condenser 116 using tube connecting blocks (not shown), which are known in the art. The receiver-dryer 118 is shown positioned beside the condenser 116, but may be positioned in front thereof to maximize the efficiency of the refrigerant by using cooling fins as shown in Figure 3. The unique design and construction of the receiver-dryer 118 will be discussed in more detail below with regard to Figures 3 and 4.

[0040] The following discussion will refer simultaneously to the apparatus of Figure 1 and to the graphical depiction of the function of that apparatus in Figure 2. Referring to Figure 1, the refrigeration cycle continues within the IRDC 114 to change the pressurized refrigerant fluid from its gaseous phase to a liquid phase, as represented by path A-D' in the pressure vs. enthalpy diagram of Figure 2. Referring

to Figure 1, the superheated vapor of the refrigerant fluid flows back and forth, winding its way down through the interior of the passages 140 of the condenser 116 while ambient air flows over the cooling fins 142 and exterior surfaces 144 of the passages 140. The superheated vapor is much hotter than the ambient air and, thus, the heat of the superheated vapor is given off to the surrounding ambient air flowing over the fins 142 and other exterior surfaces 144 of the condenser 116, thereby cooling the refrigerant fluid in accord with heat transfer principles. In other words, as the superheated vapor of the refrigerant fluid continues to flow through the condenser 116 and lose more heat to the surrounding ambient air, it begins to condense from its high pressure superheated gaseous phase into a high pressure liquid phase. Point B in the pressure vs. enthalpy diagram of Figure 2 corresponds to a location in the condenser 116 of Figure 1 that likely varies between the inlet port 146 and the second separator D2.

[0041] Similar to prior art Figures 5 and 6, point X of Figure 1 corresponds to the variable range X depicted in Figure 2, wherein the refrigerant exiting the intermediate outlet port 150 is predominantly a liquid phase but also includes some gaseous phase as a result of the cooling capacity. Like the previous discussion with reference to Figure 6, here range X in Figure 2 represents the liquid and gaseous phase of the refrigerant fluid at an intermediate portion of the condenser 116 at point X in Figure 1. Whereas point A in Figure 2 is well defined and fixed at the location on the pressure vs. enthalpy diagram as shown, any one point within range X is not so well-defined and varies along the condenser path B-C (146 to 150 and from 166-148) of the pressure vs. enthalpy diagram depending upon the vehicle performance variables of vehicle speed and load on the air-conditioning system as illustrated in

Figure 1 from reference character 146 to 150 and 166 to 144. The slower the vehicle speed and at idle, and the higher the load on the air-conditioning system, any one point within the range X will move in the direction of point B. In other words, the point within range X can vary from a saturated vapor to a sub-cooled liquid or anywhere in between such as a liquid-vapor mixture. In contrast to the prior art system and diagram of Figures 5 and 6, here with the system and diagram of Figures 1 and 2 of the present invention, point D' is providing additional amounts of sub-cooling that can be performed within the system Y_2.

[0042] Rather, point D' is also influenced by the ability of the present invention to provide subsequent efficient sub-cooling and separation of liquid and gas phases of the refrigerant fluid beyond point $X+Y_1$ (between point X and point Y_1) and further subsequent sub-cooling beyond point Y_1 to point Y_2. As shown in Figure 1, the receiver-dryer 118 is a vertically disposed vessel for separating the refrigerant wherein the mixture of gaseous-liquid phase rises to the top of, and captures the gaseous phase within the upper portion 158 thereof, yet the liquid phase of the refrigerant falls under gravity and settles in the lower portion 164 thereof. Accordingly, location Y in Figure 1 corresponds to the sub-cooling range Y_1+Y_2 depicted in Figure 2, wherein the refrigerant entering the intermediate inlet port 166 of the condenser 116 is saturated or sub-cooled liquid refrigerant (point C). The refrigerant at location C is mostly saturated liquid refrigerant at location X, because the refrigerant at location X is a varying combination of liquid and gaseous phases whereas the refrigerant at location Y (166) is a stable supply of liquid phase separated in the bottom chamber or outlet line 162 of the receiver dryer 164. Additional sub-cooling takes place within the condenser 116 between the intermediate inlet port or

point 166 and the outlet port 148 whereat the pressurized sub-cooled refrigerant fluid exits the condenser 116 at Point D' as a liquid phase, flows downstream through the refrigerant line 124', and enters the thermal expansion valve 120.

[0043] Accordingly, the present invention ensures the presence of sub-cooling and increases the magnitude thereof. This can best be seen by comparing the leftward shift of line D'-F' of Figure 2 compared to the position of line D-F of prior art Figure 6. In other words, the present invention increases the enthalpy range from point O to point F' as seen in Figure 2, compared to the prior art enthalpy range from point O to point F of Figure 6. The amount of heat (Q) that can be removed by the present invention air-conditioning system is represented by the equation $Q=M_{R134a}*(h2-h1')$. M_{R134a} is the variable mass flow for R134a refrigerant while h2 is the enthalpy at the end of the compression cycle O-A and h1' is the enthalpy at the end of the condensing cycle A-D'. Assuming a constant mass flow, the greater the range in enthalpy that the air-conditioning system can produce, the greater the heat that can be removed. Therefore, by increasing the enthalpy range compared to the prior art, the present invention thereby increases the amount of heat that can be removed from the refrigerant fluid, which translates to an increase in efficiency of the present invention air-conditioning system compared to the prior art.

[0044] Continuing through the refrigeration cycle, and referring to Figure 1, the thermal expansion valve 120 may be any type of adiabatic expansion device that "expands" the condensed high pressure refrigerant liquid so as to suddenly reduce the pressure of the refrigerant liquid to a low pressure liquid and gas phase mist. This sudden reduction in pressure causes the refrigerant fluid to be sprayed through the refrigerant line 124' downstream to the evaporator 122. The opening of the thermal

expansion valve 120 is controlled by a thermostat 168 located downstream of the evaporator 122 for maintaining a constant temperature of the refrigerant exiting the evaporator 122. This process is represented in the Figure 2 pressure vs. enthalpy diagram by path D'-F', wherein point E' represents the point at which the refrigeration cycle crosses the saturated liquid line such that the refrigerant changes from a sub-cooled liquid to a saturated liquid. Point F' represents the liquid/gas phase refrigerant in a fully expanded state ready for evaporation.

[0045] Referring again to Figure 1, the evaporator 122 is positioned downstream of the thermal expansion valve 120 and is preferably located within a passenger compartment of the motor vehicle such as under an instrument panel thereof (not shown). The evaporation process extracts the required latent heat from an incoming stream of fresh or recirculating air by way of a blower (not shown), thereby cooling the air. Within the evaporator 122, the now depressurized liquid phase of the refrigerant fluid changes back into a gaseous phase. While the now relatively cool refrigerant fluid flows through interior passages of the evaporator 122, relatively hot ambient air flows over exterior surfaces of the evaporator 122. The evaporator 122 cools and dehumidifies the hot moist ambient air, because the humidity or water vapor in the hot moist ambient air collects or condenses on the exterior of the evaporator 122. The evaporator 122 also cools the hot moist ambient air because the heat of the hot moist ambient air is given off to the relatively cold refrigerant flowing through the evaporator 122, thereby warming the refrigerant fluid and cooling the air flowing over the exterior surfaces of the evaporator 122. Thus, a supply of cool and dehumidified conditioned air flows away from the evaporator 122 and into the passenger compartment of the motor vehicle, while the evaporated gaseous refrigerant flows out

of the interior passages of the evaporator 122, through the refrigerant line 124" downstream back to the compressor 112 where the refrigeration cycle repeats. This process is represented in the Figure 2 pressure vs. enthalpy diagram by path F'-O, wherein point G represents the point at which the refrigerant changes from a saturated liquid-gas mixture to a saturated gas. The cycle illustrated in Figure 2, OA to AD' to D'F' to F'O is transient in nature with vehicle speed and ambient heat load.

[0046] Figure 3 illustrates an enlarged view of the receiver-dryer 118 shown in Figure 1. The housing 154 is preferably composed of a thin-walled metal such as a 6063-T6 aluminum alloy, but may be composed of other aluminum, steel, plastic, and the like. The inlet and outlet tubes 152, 162 are preferably brazed to the housing 154 and are preferably composed of a 3003-H14 aluminum alloy, but may be composed of other aluminum, steel, plastic, and the like. The receiver-dryer 118 of Figure 1 is a substantially cylindrical vessel, container, or housing having a base wall 170, a side wall 172 extending vertically upwardly from the base wall 170, and a concave end 174 terminating the side wall 172. The concave end 174 need not, but may, take the form of a thin-walled spherical wall, just as long as a concave interior surface is defined by the concave end 174. The walls 170, 172, 174 collectively define an interior of the receiver-dryer housing 154. The refrigerant inlet pipe 152 extends into the interior of the housing 154 and terminates in the exit end 156 facing the concave interior surface of the concave wall 174 of the housing 154. The receiver-dryer 118 also includes the integrated filter and adsorbent unit 160 that is centrally disposed over the inlet tube 152 and that is elevated by one or more indentations 176 formed into the side wall 172 of the housing 154. The unit 160 may be a saddle bag type device, a puck-like device, or any other suitable desiccant and filter device that is

known. The unit 160 effectively divides the interior of the housing 154 into the upper portion 158 above the unit 160 and the lower portion 164 below the unit 160.

[0047] The inlet tube 152 is adapted for directing the refrigerant fluid into contact with the concave end wall 174 such that the refrigerant fluid impinges on the inner concave end wall 174 to separate the mixture of liquid/gaseous refrigerant fluid into a gaseous phase that accumulates in the upper portion 158 of the housing 154 and a liquid phase that by adhering to the interior concave end wall falls under gravity to accumulate in the lower portion 164 of the housing 154. The design of the concave wall 174 and proximity of the exit end 156 of the inlet tube 152 is adapted for substantial contact of liquid refrigerant and relatively uniform dispersion of refrigerant so that a substantial amount of refrigerant liquid adheres to the inner surfaces of the housing 154 due to liquid surface tension and wherein the liquid runs down interior surfaces of the concave wall 174 and side wall 172 for heat transfer cooling therewith. Additional efficiency maybe obtained by the use of cooling fins 178 as shown in Figure 3. Therefore, cooling fins 178 are preferably disposed on the exterior of the housing 154 for increased heat transfer cooling of the refrigerant fluid. The combined secondary surface area of the fins 178 is represented by element A_S and the combined primary surface area of the concave wall 174 and side wall 172 in the upper portion 168 of the housing 154 is represented by element A_p. According to the present invention, A_S is preferably greater than A_p. The unique design of the concave wall 174 and proximity of the inlet tube 152 with respect thereto enables relatively greater dispersion of the refrigerant fluid, and the cooling fins 178 enable relatively greater conversion of the refrigerant fluid into a liquid phase. Both features provide for greater condensing of the refrigerant gas phase into liquid phase. The fins

178 may be separately attached to the housing 154 such as by brazing, or may be assembled thereto as a separate sub-assembly. In a similar vein, Figure 4 illustrates an alternative embodiment of the present invention, in which the heat transfer functionality of the cooling fins is substituted by an isomount hat 180 or maybe integrated with the cooling fins.

[0048] The isomount hat 180 includes a socket shaped portion 182 that is adapted for heat transfer contact with the top of the housing 154 and further includes a bracket portion 184 that is adapted for fastening to another structural member such as the condenser 116 or any other proximate structure within an engine compartment. Accordingly, the top of the receiver-dryer 118 may be firmly supported and mounted within the engine compartment for less vertical and lateral movement of the receiver-dryer 118. The socket shaped portion 182 is concave shaped for conforming contact with the convex shaped concave wall 174 of the housing 154. The socket shaped portion 182 is also preferably constructed of a relatively high thermally conductive material such as aluminum or steel and may have a metallic or non-metallic outer skin. It is contemplated that the isomount hat 180 could be used in combination with the cooling fin arrangement of Figure 3. In any case, a secondary surface area As' should be greater than the primary surface area Ap.

[0049] Referring again to Figure 3, the outlet tube 162 has an entrance end 186 in fluidic communication with the lower portion 164 of the housing 154 for permitting only the liquid phase of the refrigerant and a lubricant to exit the receiver-dryer 118. The level of saturated liquid and lubricant will change depending upon the condensing capacity of the apparatus, the cooling load placed on the refrigeration system, vehicle performance, and the like.

[0050] The receiver-dryer 118 may be manufactured according to any of the well-known techniques for forming aluminum canisters, but is preferably constructed by the following described process. The housing 154 preferably originates from tube stock which is impact closed to form the flat bottom end or base wall 170. However, the housing 154 may originate from sheet or tubular stock, which is then deep drawn to form the base wall 170. Holes are then drilled in the closed bottom end or base wall 170 and the inlet and outlet tubes 152, 162 are inserted therein and brazed to the housing 154. The inlet tube 152 is inserted within the housing 154 such that the exit end 156 thereof faces the top inside surface of the concave wall 174 and is disposed within a distance that is substantially proximate the radius of the spherical-shaped concave wall 174 of the housing 154. Alternatively, the exit end 156 may be spaced from the top inside surface within proximity of the radius dimension of the spherical concave wall 174. Then, the indentation(s) 176 are formed in the side wall 172 of the housing 154 by tri-crimping or forming cylindrically the housing 154, or the like. Next, the integrated filter and adsorbent unit 160 is assembled into the interior of the housing 154. The open end of the tube stock is spun closed to form the closed top end or concave interior wall 174. Spin closing of aluminum containers is generally known in the art, e.g. by U.S. Pat. No. 5,245,842, which is incorporated by reference herein. Uniquely, however, the top end or concave wall 174 is preferably spun closed in such a manner so as to achieve a concave, rounded, and preferably spherical, top inside surface of the concave wall 174.

[0051] In accordance with the present invention, the preferred method involves improved sub-cooling of the refrigerant within an air conditioning system. The method may be practiced in accord with the air conditioning system 110 of

Figure 1, but may also be practiced using any suitable air conditioning system. The method includes receiving a superheated gaseous phase of a refrigerant fluid in a condensing stage of a condenser, and condensing the superheated gaseous phase of the refrigerant fluid within the condensing stage into a mixture of a gaseous phase and a liquid phase of refrigerant. The method further involves communicating the mixture into a vertically disposed container, housing, or vessel, and directing the mixture into a top concave surface of the vertically disposed container, thereby dispersing the liquid phase from the gaseous phase wherein the liquid phase falls toward a lower portion of the container over a desiccant material, and further thereby cooling the gas and liquid phases for improved sub-cooling of the liquid phase by adhering to the interior concave wall 174 and for improved condensing of the gas phase into the liquid phase. Accordingly, the method produces a separated, cooled, and dehydrated liquid phase that accumulates in the lower portion of the container. Finally, the method includes communicating the separated, cooled, and dehydrated liquid phase out of the container and back into a sub-cooling stage of the condenser.

[0052] With each of the embodiments described above, a condenser stage of a refrigeration cycle is optimized for greater dispersion and increased cooling of refrigerant to condense a relatively greater amount of gaseous phase refrigerant into liquid phase refrigerant. The present invention thereby provides for increased sub-cooling of the refrigerant for cooler air output in a passenger compartment of an automobile per a given work input of a compressor, thereby increasing the efficiency of the air conditioning system.

[0053] While the present invention has been described in terms of a preferred embodiment, it is apparent that other forms could be adopted by one skilled in the art.

In other words, the teachings of the present invention encompass any reasonable substitutions or equivalents of claim limitations. For example, the structure, materials, sizes, and shapes of the individual components could be modified, or substituted with other similar structure, materials, sizes, and shapes. Specific examples include providing slight alterations to the shape of the concave end of the receiver-dryer vessel that achieve similar beneficial results as the present invention. Those skilled in the art will appreciate that other applications, including those outside of the automotive industry, are possible with this invention. Accordingly, the present invention is not limited to only automotive refrigeration systems. Accordingly, the scope of the present invention is to be limited only by the following claims.

What is claimed is: